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Comparison between single and cascaded organic Rankine cycle systems accounting for the effects of expansion volume ratio on expander performance

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Abstract. Compared to single-stage organic Rankine cycle (ORC) systems, cascaded ORC systems, in which a high-temperature topping cycle and low-temperature bottoming cycle are coupled together, could have advantages in terms of removing the potential for sub-atmospheric condensation conditions and improving expander performance as the expansion process is effectively divided across two stages. Moreover, reducing the expansion volume ratio could facilitate the use of volumetric expanders, such as twin-screw expanders, which, in turn, could facilitate two-phase expansion to be utilised in one, or both, of the cycles. The aim of this paper is to compare single-stage and cascaded ORC systems, accounting for the effect of the expander volume ratio on expander performance. To investigate this, thermodynamic models for single-stage and cascaded ORC systems are developed, which include variable efficiency expander models for both radial turbines and twin-screw expanders that can estimate the effect of the expansion volume ratio on the expander isentropic efficiency. Using this model, three different scenarios are compared for different temperature heat-source temperatures, namely: (i) single-stage ORC systems with vapour-phase expansion obtained using a turboexpander; (ii) single-stage ORC systems operating with a twin-screw expander, with the possibility for two-phase expansion; and (iii) cascaded cycles with either vapour- or two-phase expansion. The results from this comparison are used to identify applications where cascaded ORC systems could offer performance benefits.

1. Introduction

Power generation systems based on organic Rankine cycles (ORC) are widely considered to be one of the most promising technologies for power generation from low-temperature heat, typically below 400 °C. At the domestic and commercial scales (*i.e.*, less than a few hundred kilowatts), most commercial systems are based on a single-stage system constructed from a pump, evaporator, expander and condenser. However, for heat-source temperatures at the higher end of the range, the resulting systems are typically associated with sub-atmospheric condensation pressures, which leads to physically large condensers that need to operate under a vacuum, and are also associated with large volumetric expansion ratios.

Large volumetric expansion ratios have implications on both expander selection and design. Whilst radial turbines are capable of obtaining the required expansion over a single stage, it is likely that the turbine efficiency will be reduced owing to small blade heights, which introduce increased secondary flows and clearance losses [1]. On the other hand, volumetric expanders, such as scroll or screw expanders, are limited in their volume ratio by mechanical design constraints, such as thermal distortion and bearing loads, and thus can only be considered if multiple expansion stages are used. However, even then, the relatively low volumetric expansion ratio achievable within each stage may limit the overall volumetric expansion ratio to 20 for two-stage expansion [2]. Moreover, the large volumetric expansion ratios also rule out the potential for two-phase expansion, which has been theoretically shown to increase power output from waste-heat recovery systems [2, 3, 4], as two-phase expansion cannot be realised using turbo-expanders.

In comparison, cascaded systems, in which a high-temperature topping cycle and low-temperature bottoming cycle are coupled together, could offer performance benefits over the single-stage system. Firstly, the expansion process is effectively divided over two-stages, and thus the volumetric expansion ratios within both the topping and bottoming cycles are reduced. This introduces the possibility to design more efficient turbines, and the possibility to use volumetric expanders, and most importantly the possibility to realise two-phase expansion in one, or both, of the cycles. Moreover, since both the topping and bottoming cycles represent individual closed loops it is possible to select different working fluids for each cycle. Thus, fluids can be selected to effectively remove sub-atmospheric condensation pressures in both cycles, which could lead to more compact heat exchangers. A few early studies have demonstrated the potential of cascaded ORC systems for low-temperature systems below 200 °C [5, 6]. More recently, the authors have presented preliminary investigations into the optimisation of cascaded ORC systems for higher temperature heat sources, and found that both cascaded and single-stage systems can produce similar power outputs [7]. However, this previous study was limited in that only single-phase superheated expansion was considered, and that constant expander efficiencies were assumed and thus there was no consideration of the effect of the cycle conditions on expander performance.

The aim of this current paper is to complete a more detailed comparison between cascaded and single-stage systems. More specifically, the potential to use two-phase expansion in either the topping, or bottoming, cycles is introduced, whilst expander models for both radial turbines and twin-screw expanders are introduced to account for the effect of the volumetric expansion ratio on expander efficiency. Using this model allows a more rigorous comparison of the different types of cycle, and can thus help identify optimal solutions for waste-heat recovery applications.

2. Description of the model

2.1. Thermodynamic model

The notation used to describe the single-stage and cascaded ORC systems is described in Figures 1 and 2 respectively. For both systems, it is assumed that the system is operating under steady-state conditions, whilst heat losses and pressure drops within the pipework are neglected. Fluid properties are accounted for using NIST REFPROP. For each cycle, the cycle analysis is completed by calculating all of the state points within the cycle based on an input vector of cycle variables, and then applying energy balances to each heat exchanger within the cycle to determine the working-fluid mass-flow rate and heat-source or heat-sink outlet conditions. For brevity, the precise details of these calculations are not described here, but details can be found in the authors' previous work [7, 8]. The pump is modelled using a fixed isentropic efficiency, whilst the expander is modelled as described in Sections 2.2 and 2.3.

For a single-stage system, there are four variables that can be optimised to maximise the power output from this system. This can be expressed as:

$$\max\{\dot{W}_{\mathrm{n}} = f(T_1,\ p_r,\ PP_{\mathrm{h}},\ q_3)\}\,, \tag{1}$$

where $\dot{W}_{\mathrm{n}}$ is the net-power output from the cycle, T_1 is the condensation temperature, p_r is the reduced evaporation pressure (p_2/p_{cr}, where p_{cr} is the fluid critical pressure), and PP_{h} is the pinch-point temperature difference at the beginning of evaporation. The final variable, q_3, defines the expander inlet conditions and is defined in such a way as to allow either two-phase

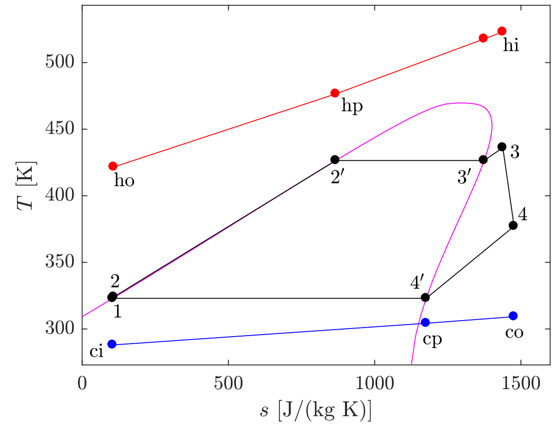

Figure 1. Temperature-entropy (T-s) plot of a single-stage ORC system.

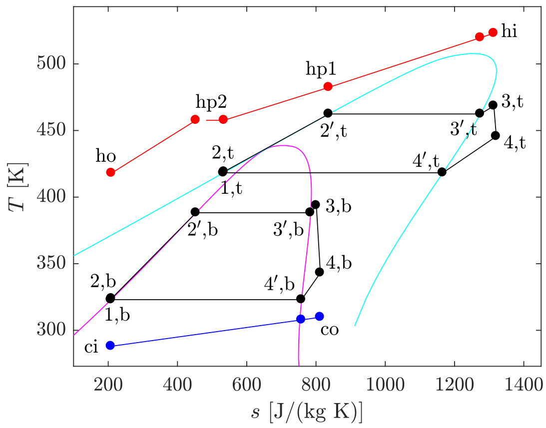

Figure 2. Temperature-entropy (T-s) plot of a cascaded ORC system.

or single-phase expansion. For $q_3 \leq 1$, q_3 is equal to the vapour quality at the expander inlet, whilst for $q_3 > 1$, the expander inlet temperature is found from:

$$T_3 = T_{3'} + (q_3 - 1)(T_{\mathrm{hi}} - T_{3'})\,, \tag{2}$$

where $T_{3'}$ is the evaporation saturation temperature and T_{hi} is the heat-source inlet temperature.

For a cascaded system, there are seven variables that can be optimised to maximise the performance of the system:

$$\max\{\dot{W}_n = f(T_{1,\mathrm{b}},\ p_{r,\mathrm{b}},\ p_{r,\mathrm{t}},\ q_{3,\mathrm{t}},\ PP_{\mathrm{h,t}},\ \Delta T_{\mathrm{sat}},\ T_{\mathrm{ho}})\}\,. \tag{3}$$

In this case, the first five variables have the same meaning before, but with the subscripts ‘t’ and ‘b’ introduced to refer to the topping and bottoming cycles respectively. The sixth optimisation variable describes the temperature difference between the bottoming-cycle evaporation temperature and the topping-cycle condensation temperature (*i.e.*, $\Delta T_{\mathrm{sat}} = T_{1,\mathrm{t}} - T_{3',\mathrm{b}}$), whilst T_{ho} defines the heat-source outlet temperature. For the cascaded cycle, it should be noted that the expander inlet conditions within the bottoming cycle are not directly controlled, but instead result from an energy balance applied to the intermediate heat exchanger that transfers heat from the topping cycle to the bottoming cycle. Further details on the cascaded-cycle model can be found in Ref. [7].

2.2. Turboexpander model

For small-scale ORC applications of the order of a hundred kilowatts and below radial turbines are the most common type of turboexpander selected. Broadly speaking, the efficiency of a radial turbine will be dependent on two factors, namely the volumetric expansion ratio, and the turbine size. The latter is predominantly determined by the mass-flow rate, and thus depends on the power rating of the system. The former is a more fundamental effect of the thermodynamic cycle conditions, with high volumetric expansion ratios (*i.e.*, large changes in density across the turbine) resulting in a reduction in turbine efficiency. As the intention behind the current paper is compare radial turbines and screw expanders, and at present simple methods to account for scaling effects in screw expanders are not available, scaling effects within radial turbines will not be considered. Instead, the focus will be on the effect of the volumetric expansion ratio.

Perdichizzi and Lozza [9] developed a map of radial turbine efficiency as a function of the isentropic volumetric expansion ratio ($V_{r,\mathrm{s}} = \rho_3/\rho_{4\mathrm{s}}$, where $\rho_{4\mathrm{s}}$ is the density following an isentropic expansion) and the size parameter (Figure 3a). The size parameter is defined as:

$$SP = \frac{\sqrt{\dot{V}_{4s}}}{\Delta h_s^{1/4}}\,, \tag{4}$$

where $\dot{V}_{4s}$ is the expander outlet volumetric-flow rate following an isentropic expansion ($\dot{V}_{4s} = \dot{m}/\rho_{4s}$) and Δh_s is the isentropic enthalpy drop across the turbine. More recently, a similar map was has been obtained using mean-line modelling methods [10].

Using this map, the variation in turbine efficiency with $V_{r,s}$ can be evaluated at different size parameters by taking vertical slices through the contour (*e.g.*, for a size parameter of 0.03, the efficiency is 0.88, 0.87 and 0.86 at $V_{r,s} = 2$, 4 and 6 respectively *etc.*). Repeating this for different size parameters, and then normalising each set of results by the maximum efficiency for that set, the data reported in Figure 3b is obtained. Applying a linear regression to this leads to:

$$\frac{\eta}{\eta_{max}} = -0.004615 V_{r,s} + 1.007\,, \tag{5}$$

which corresponds to $R^2 = 0.9328$. Thus, neglecting any scaling effects and assuming a maximum efficiency of $\eta_{max} = 0.89$, the efficiency of the turbine operating for a particular volumetric expansion ratio can be estimated using Equation 5.

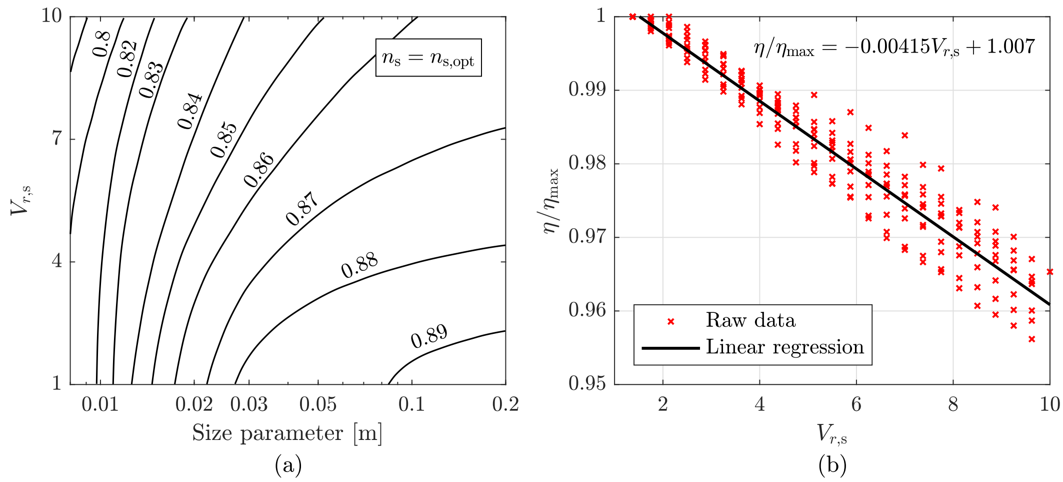

Figure 3. Turbine efficiency maps: (a) performance map developed by Perdichizzi and Lozza [9]; (b) extracted relationship between turbine efficiency and volumetric expansion ratio.

2.3. Twin-screw expander model

An alternative to a radial turbine is a positive-displacement expander, such as a twin-screw expander. Twin-screw expanders are generally used in applications with smaller volumetric expansion ratios than radial turbines, since they are limited by the built-in volume ratio of the machine. However, they have the advantage of being able to accommodate two-phase expansion and thus offer the potential to improve power output in waste-heat recovery applications. The authors have previously developed a model to account for the effect of the volumetric expansion ratio on the efficiency of a twin-screw expander [11], which will be briefly described here.

When considering a particular twin-screw expander for a defined application the two parameters of primary interest are the volumetric expansion ratio ($V_r = \rho_3/\rho_4$), and the expander built-in volume ratio ($V_{r,bi}$), which is the ratio of the chamber volume at the inlet and outlet of the machine and this is fixed for a given machine. Previously, Read et al. [12] have shown that the efficiency of a twin-screw expander is related to the ratio of these volume ratios, defined as $R_{\text{exp}} = V_{r,\text{bi}}/V_r$. This relationship is shown in Figure 4, and it is observed that an optimal expander efficiency is obtained for $R_{\text{exp}} \approx 0.65$. The maximum built-in volume ratio for a twin-screw expander is limited by mechanical design constraints; as the required built-in volume ratio increases, the length of the rotor increases, which introduces challenges with regards to thermal distortion and bearing loads. Within this paper, it is assumed that the maximum built-in volume ratio is 5. Therefore, the maximum volumetric expansion ratio that can be achieved using a twin-screw expander, without resulting in a reduction in the expander isentropic efficiency, is $5/0.65 = 7.7$. Thus, it is possible to derive an expression for the expander isentropic efficiency as a function of the volumetric expansion ratio. For $V_r \leq 7.7$ it is assumed that a twin-screw expander can be selected, or designed, that has an optimal built-in volume ratio such that $R_{\text{exp}} = 0.65$. For $V_r > 7.7$, the expander isentropic efficiency can be estimated using a second-order polynomial curve fit for the data shown in Figure 4. This is expressed as follows:

$$\eta = \begin{cases} \eta_{\text{max}}, & \text{if } V_r \leq 7.7 \\ -0.7205R_{\text{exp}}^2 + 0.9230R_{\text{exp}} + 0.5100, & \text{otherwise} \end{cases} \tag{6}$$

where $\eta_{\text{max}} = 0.806$. This correlation is shown in Figure 5.

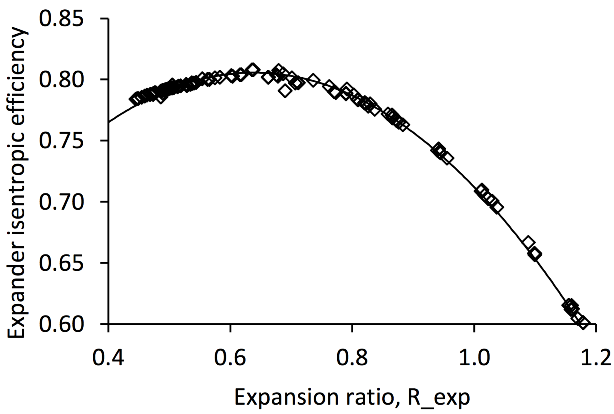

Figure 4. Relationship between the expansion ratio ($R_{\text{exp}} = V_{r,\text{bi}}/V_r$) and the isentropic efficiency of a twin-screw expander [12].

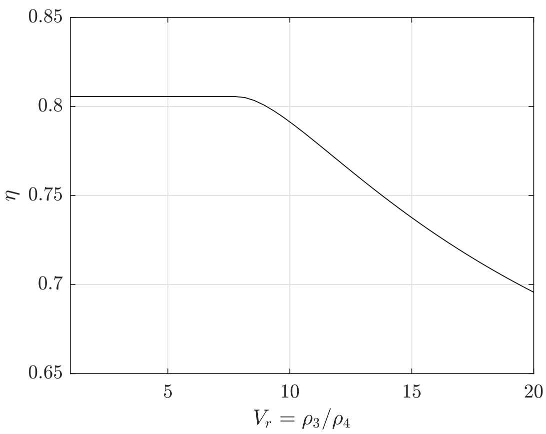

Figure 5. Twin-screw expander efficiency as a function of expansion volume ratio V_r, assuming a maximum built-in volume ratio of 5.

2.4. Optimisation setup

The combination of the two cycle models described in Section 2.1 with the two expander models described in Sections 2.2 and 2.3 facilitates a thorough investigation into single-stage and cascaded ORC systems with and without two-phase expansion. In total, six different cycle configurations can be considered and these are summarised in Table 1.

Within this study, the six different cycles will be compared for three different heat-source temperatures, namely $T_{\text{hi}} = 473$, 523 and 573 K. As the expander efficiencies are modelled based only on the volumetric expansion ratio, and thus effects related to the size of the machine are not considered, a heat source of hot air with an arbitrary mass-flow rate of 1 kg/s is defined. The heat sink is assumed to be water at a temperature of 15 °C and is also defined with a mass-flow rate of 1 kg/s. The pumps are modelled with a fixed isentropic efficiency of 70%, whilst a constraint is applied to each heat exchanger to ensure that the minimum allowable temperature difference exceeds 10 K. The bounds for the optimisation parameters are summarised in Table 2.

As reported in Table 2, seven different working fluids are considered. For cascaded systems the number of optimisation studies required for each heat-source temperature is equal to the square of the number of fluids considered. Thus, other optimal fluid candidates may exist, but to evaluate all possible fluid combinations is unfeasible. However, the authors' have previously demonstrated how heat-source temperature and critical temperature are linked for both single-stage [8] and cascaded systems [7], and thus these fluids have been selected as they are commonly considered within the literature and span a range of relevant critical temperatures. Nonetheless, future studies should investigate in more detail the effects of the fluid on the expansion process (*i.e.*, wet or dry) and how this affects the results of the optimisation.

Each optimisation is completed in MATLAB using the sequential quadratic programming algorithm within the Optimisation Toolbox (MATLAB 2017a, The Mathworks, Inc.). For the single-stage systems, each working fluid is considered in turn, whilst for the cascaded systems every possible pairing of the seven working fluids is considered. Each optimisation is completed from 10 different start points to help ensure a global optimum is identified. Thus, for each combination of cycle and heat-source temperature, a total of 70 optimisations are completed for the single-stage systems and 490 optimisations are completed for the cascaded cycles. In each case, the optimal cycle is the one that produces the maximum power.

Table 1. Summary of the six different cycles considered within this study.

Label	Type	Expander (top)	Expander (bottom)	Expansion (top)	Expansion (bottom)
1-T	single		turbine		single-phase
1-S	single		screw		single- or two-phase
2-TT	cascaded	turbine	turbine	single-phase	single-phase
2-TS	cascaded	turbine	screw	single-phase	single- or two-phase
2-ST	cascaded	screw	turbine	single- or two-phase	single-phase
2-SS	cascaded	screw	screw	single- or two-phase	single- or two-phase

Table 2. Bounds for the optimisation parameters and list of working fluids considered.

Variable	min.	max.	units	Fluid	T_{cr} [K]
T_1, $T_{1,\mathrm{b}}$	298	373	K	isobutane	407.8
p_r, $p_{r,\mathrm{b}}$, $p_{r,\mathrm{t}}$	0.05	0.85	–	R245fa	427.2
q_3, $q_{3,\mathrm{t}}$	0 (screw)	2	–	R1233zd	438.8
	1 (turbine)	2	–	isopentane	460.4
PP_{h}, $PP_{\mathrm{h,t}}$	10	100	K	n-pentane	469.7
ΔT_{sat}	10	100	K	cyclopentane	511.7
T_{ho}	288	T_{hi}	K	benzene	562.0

3. Results

The maximum net power output obtained for each cycle and each heat-source temperature are summarised in Figure 6 and Table 3. Considering the single-stage systems (1-T and 1-S), it is found that the 1-S systems result in between 10% and 15% less power than the 1-T systems. This can be partly attributed to twin-screw expanders being associated with inherently lower efficiencies than turbines, but also the result of the cycle operating conditions which have an impact on expander performance. Firstly, screw efficiency drops more drastically with an increase in volumetric expansion ratio than turbine efficiency (see Figures 3 and 5). Thus, as the heat-source temperature increases, the 1-T systems can accommodate higher volume ratios more effectively than the 1-S systems, which maintain reasonable screw efficiency at the expense of limiting the volume ratio and thus limiting the performance of the cycle. This is further confirmed by considering the volume ratios for the single-stage systems (Figure 7), which show that for the 573 K heat source the volume ratio for the screw system is almost half of the volume ratio for the equivalent turbine system.

Table 3. Summary of the optimal cycles identified for each cycle and heat-source temperature.

	$T_{hi} = 473$ K			$T_{hi} = 523$ K			$T_{hi} = 573$ K		
	$\dot{W}_n$	bottom	top	$\dot{W}_n$	bottom	top	$\dot{W}_n$	bottom	top
1-T	17.4	R245fa	–	27.3	pentane	–	38.2	cyclopentane	–
1-S	15.7	isopentane	–	24.3	R1233zd	–	32.6	cyclopentane	–
2-TT	16.7	isobutane	R1233zd	27.8	R1233zd	*n*-pentane	40.5	*n*-pentane	cyclopentane
2-TS	15.2	R245fa	R245fa	25.9	isopentane	*n*-pentane	38.4	*n*-pentane	cyclopentane
2-ST	17.2	isobutane	pentane	28.4	isopentane	benzene	40.4	cyclopentane	benzene
2-SS	16.2	isobutane	benzene	27.4	isopentane	benzene	38.6	cyclopentane	benzene

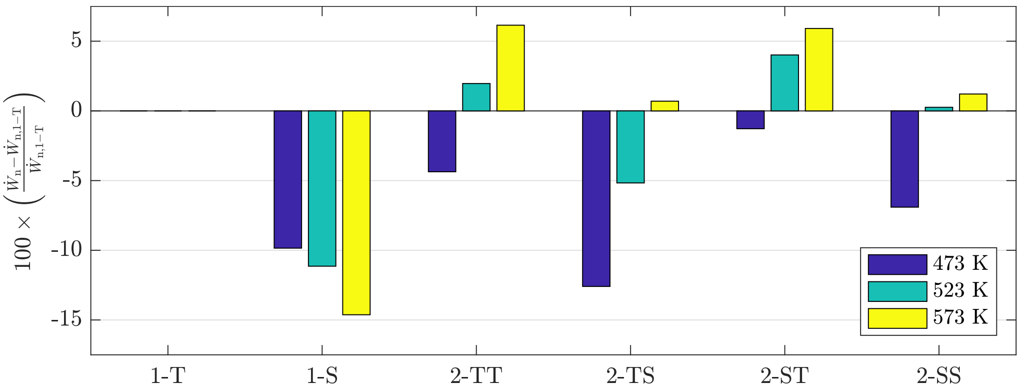

Figure 6. Comparison of the optimal power output produced for each cycle and heat-source temperature. Results are shown relative to the optimal 1-T sysem.

For the cascaded systems reported in Figure 6, it is observed that for the 473 K heat source, the power output for all of the cascaded cycles is less than the power output for 1-T systems. This is because even though expander efficiency is increased in the cascaded cycles, owing to the smaller volume ratios (Figure 7), the increased irreversibility introduced with the additional heat exchange processes is significant enough to offset these gains in expander performance. Thus, for a 473 K heat source, there is no thermodynamic benefit in operating a cascaded cycle over a 1-T system. However, as the heat-source temperature increases, a relative increase in cycle performance for the cascaded cycles is observed. This is because the trade-off between the different irreversibilities swings in the favour of the increased expander efficiencies and improved thermodynamic performance of the two cycles. This is particularly true for the two cascaded systems that utilise a turbine within the bottoming cycle (2-TT and 2-ST), which are the best performing cycles for the 523 and 573 K heat-source temperatures. These cycles are reported in Figure 8. In these cycles, the reduced volume ratios, compared to the single-stage systems (Figure 7), result in an increase in the turbine efficiency. Moreover, a significant portion of the power production within the cascaded cycles occurs within the bottoming cycle (Figure 9).

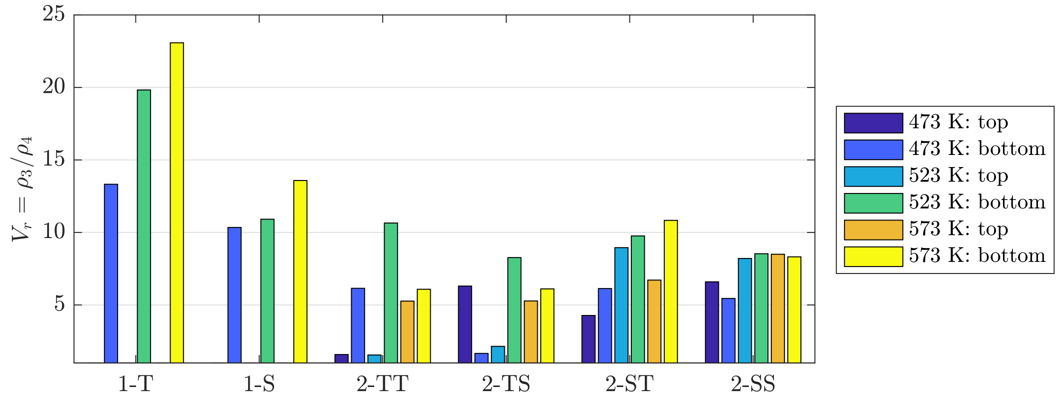

Figure 7. Volumetric expansion ratios ($V_r = \rho_3/\rho_4$) for each cycle and heat-source temperature.

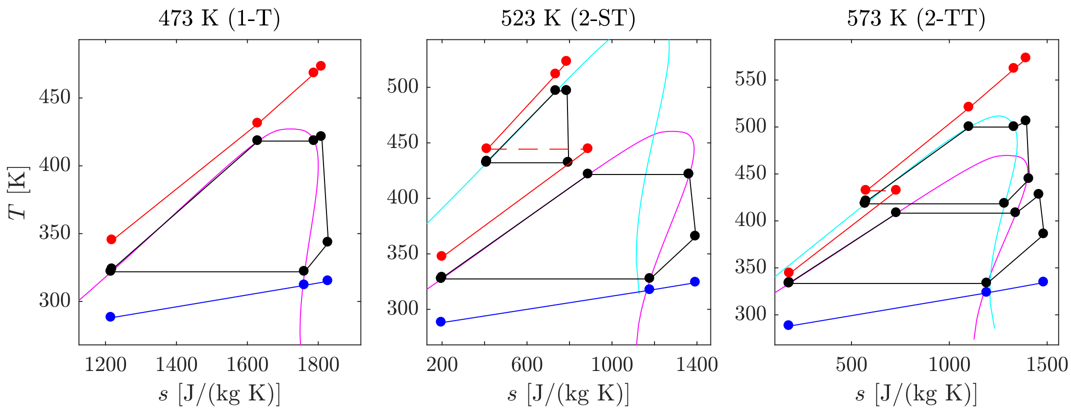

Figure 8. Optimal cycle for each heat-source temperature shown on a T-s diagram.

Thus, a large portion of the power produced is produced by a turbine that has a high efficiency. On the contrary, for the cascaded-systems with a screw in the bottoming cycle (2-TS and 2-SS), the majority of the power is still produced in the bottoming cycle, but the efficiency of the expander is lower, and hence thermodynamic performance is diminished.

Overall, the 2-TT and 2-ST systems appear the most promising candidates for the 523 and 573 K heat sources. For the 573 K heat source, the performance of both systems is similar with the 2-TT and 2-ST systems producing 6.1% and 5.9% more power than the 1-T system, whilst for the 523 K heat source, the 2-ST system is the optimal choice producing 4.0% more power than the 1-T system. The 2-TT systems are capable of the improved performance owing to higher expander efficiencies. However, the 2-ST turbines generate a similar amount of power to the 2-TT cycles, but are operating with a less efficient expander. This can be explained by examining the expander inlet conditions (Figure 10), which reveals that the 2-ST cycles correspond to two-phase expander inlet conditions in the topping cycle ($q_{3,\mathrm{t}} < 1$). In this instance, the slight drop in expander efficiency is offset by the reduced irreversibility that is achieved by partly removing part of the isothermal heat addition process in the topping-cycle evaporator. Thus, it appears that the option of having a screw expander topping cycle operating with two-phase expansion, and a turbine bottoming cycle, appears a promising option. It is worth noting here that the possibility of two-phase expansion is only introduced by considering cascaded systems, as the heat-source temperatures are sufficiently high such that the volume ratios within the 1-S systems are too high to achieve two-phase expansion (see Figures 7 and 10).

Further to this observation, it is also worth commenting that these results are generated with

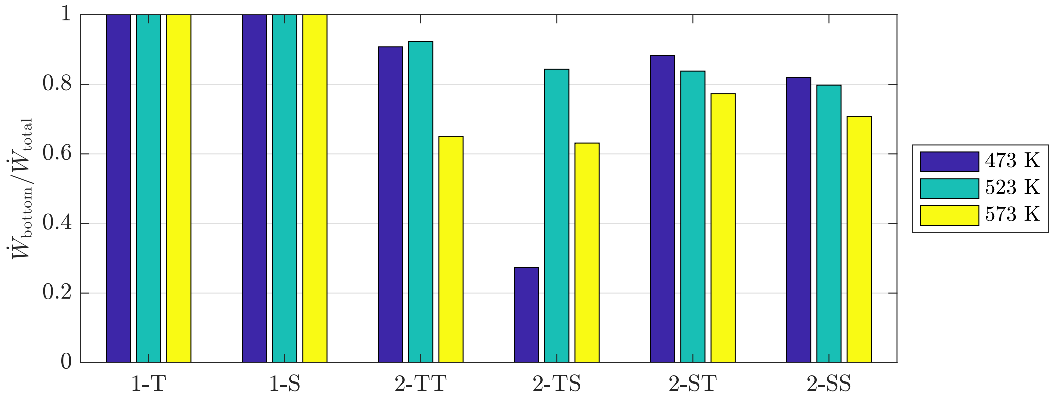

Figure 9. Ratio of bottoming-cycle power ($\dot{W}_{\mathrm{bottom}}$) to total system power ($\dot{W}_{\mathrm{total}}$).

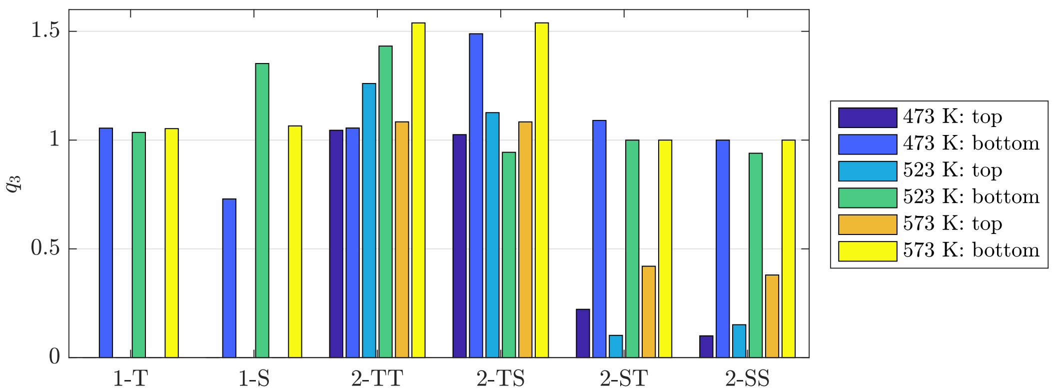

Figure 10. Optimal expander inlet conditions (two-phase inlet conditions correspond to $q_3 < 1$).

maximum efficiencies for the turbine and screw of 0.89 and 0.806 respectively. The value for the turbine was taken as the maximum from Figure 3, and scaling effects were not considered. However, in practice it is likely that this value would be lower than 0.89. For the screw expander, the maximum efficiency used is a little more conservative. In practice, it could therefore be anticipated that the 1-T and 2-TT systems would see a larger reduction in performance than the 2-ST system, which could shift the results in Figure 6 to be more in favour of the 2-ST systems. Thus, it appears that for high-temperature heat sources, the 2-ST systems appear a promising candidate and are worthy of further investigation. Future work will need to investigate the sensitivity of the results to the maximum efficiency values selected for each expander, and consider component design aspects, such as the required heat-transfer area.

4. Conclusions

Within this paper, single-stage and cascaded ORC systems has been compared by integrating expander models for radial turbines and twin-screw expanders, that account for the effect of the volumetric expansion ratio on the isentropic efficiency, with thermodynamic cycle models. Results from an optimisation study reveal that whilst for a 473 K heat-source a single-stage system operating with a turbine results in the highest power, for higher heat-source temperatures cascaded cycles generate up to 6% more power than single-stage systems. More specifically, cascaded cycles with a turbine in both cycles (2-TT), and cascaded cycles with a screw expander in the topping cycle and a radial turbine in the bottoming cycle (2-ST) appear the most promising candidates. The improved performance for these systems is due to lower volumetric expansion ratios within each cycle. For the the 2-TT systems this facilitates the design of radial turbines with higher isentropic efficiencies, whilst in the case of the 2-ST system the lower volumetric expansion ratios introduce the possibility to utilise two-phase expansion. The next steps are to extend the analysis to consider component aspects, including effects of mass-flow rate on expander efficiency and the heat-transfer area requirements for the different cycles.

Acknowledgments

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Nomenclature

η	expander isentropic efficiency
ρ	density, kg/m^3
h	enthalpy, J/kg
$\dot{m}$	mass-flow rate, kg/s
p	pressure, Pa
p_r	reduced evaporation pressure
PP_{h}	evaporator pinch point, K
q	expander inlet condition
R_{exp}	expansion ratio ($V_{r,\mathrm{bi}}/V_r$)
s	entropy, J/(kg K)
T	temperature, K
$\dot{W}_{\mathrm{n}}$	net power, W
$\dot{V}$	volumetric flow rate, m^3/s
V_r	volumetric expansion ratio
Δh_{s}	isentropic enthalpy drop, J/kg
ΔT_{sat}	saturation temperature difference, K

Subscripts

1-4	cycle state points
b	bottoming cycle
bi	built-in
ci, cp, co	heat-sink inlet, pinch and outlet
cr	critical point
hi, hp, ho	heat-source inlet, pinch and outlet
max	maximum
s	conditions after isentropic expansion
t	topping cycle